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ABSTRACT

Two-stage vapor injection compression cycle with flash tank was thermodynamically analyzed, the results showed that there existed the optimum theoretical displacement ratio of high stage to low stage corresponding to the maximum coefficient of performance(COP), the optimum displacement ratio and the volumetric heating capacity decreased with evaporation temperature decreasing. An optimum theoretical displacement ratio correlation for R290, R32 and R410A was given. A new type two-stage vapor injection low temperature air source heat pump (ASHP) was designed, which had a variable speed triple-cylinder rotary compressor with two cylinders in low stage and one cylinder in high stage. The experimental results of the new type ASHP showed that the heating capacity under 20℃/-20℃(inside room /outside room) could reach the rated heating capacity under 20℃/7℃, improving 96% compared to conventional one-stage ASHP, the heating capacity under 20℃/-30℃ could reach 80% of the rated one. COP of the new type ASHP could improve 5%~10% when the heating capacity was comparable to the conventional ASHP, and the heating capacity of the new type ASHP could improve 30%~50% when COP was comparable to the conventional ASHP.

1. INTRODUCTION

Two-stage vapor compression with vapor injection technique can improve heating capacity as well as coefficient of performance (COP) of low temperature air source heat pump (ASHP) in cold climate. Heating capacity and COP of the two-stage vapor injection ASHP were improved up to 30%~40% and 20% (Wang, 2009 & Shen, 2015), respectively, as compared to the baseline single-stage ASHP. Redón (2014) analyzed the optimum parameters, such as displacement ratio and intermediate pressure, corresponding to the maximum COP for two-stage vapor injection heat pump system with flash tank as well as economizer, refrigerants including R407C, R22, R290 and R32. A quasi two-stage scroll compressor (Wang, 2009) or a two-stage rotary compressor with two cylinders (Heo, 2011 & Xu 2014) was mainly used into ASHP in cold climate region in practice.

Compared to two scroll or rotary compressors in series, a quasi two-stage scroll or two-stage rotary compressor would be lower expense and could avoid potential oil return risk. However, it is not good enough both for heating capacity and COP of a two-stage vapor injection ASHP operating in a broad range ambient temperature in cold or extremely cold climate region since a two-stage compressor usually has only one constant displacement ratio.

This paper analyzed the optimum theoretical displacement ratio, COP and the volumetric heating capacity through thermodynamic model for two-stage vapor injection compression cycle with flash tank. Enhancement percentage of COP and volumetric heating capacity of two-stage vapor injection compression cycle compared to those of one stage vapor compression cycle under different conditions were discussed. The correlation of optimum theoretical displacement ratio was given. A new type two-stage vapor injection low temperature ASHP with a variable displacement ratio rotary compressor was designed to improve both capacity and COP. Experimental investigate with R410A as a refrigerant was carried out and tested results were concluded.

2. THEORETICAL ANALYSIS OF TWO-STAGE VAPOR INJECTION CYCLE

2.1 Two-stage Vapor Injection Cycle with Flash Tank

The schematic figure of one stage cycle and two-stage vapor injection cycle with flash tank are shown in figure 1 and figure 2, respectively. The p-h diagrams of the former cycle and the latter cycle are shown in figure 3, in dashed line and real line, respectively. As is shown in the three figures, the main differences between the two cycles are as following:

(1) Low pressure refrigerant vapor coming from evaporator is compressed twice through compressors of low and high stage, and high pressure refrigerant liquid is throttled twice through expansion valves of first and second stage.

(2) The first throttled two-phase refrigerant is separated in flash tank, the nearly saturated vapor phase is injected through injection port and mixed with discharge vapor from low-stage compressor, and then the mixed vapor enters suction port of high-stage compressor, which will reduce the suction and discharge temperature of high-stage compressor. The nearly saturated liquid is second throttled and enters the evaporator, reducing the inlet enthalpy of the evaporator.

(3) The total compression ratio is shared between low-stage compressor and high-stage compressor, reducing compression ratio of each stage.

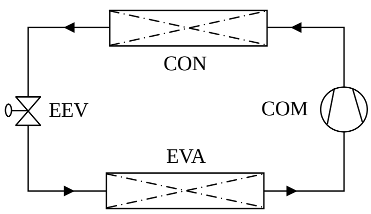


Figure1: One stage cycle

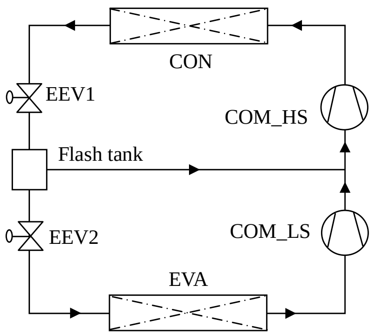


Figure 2: Two-stage vapor injection cycle with flash tank

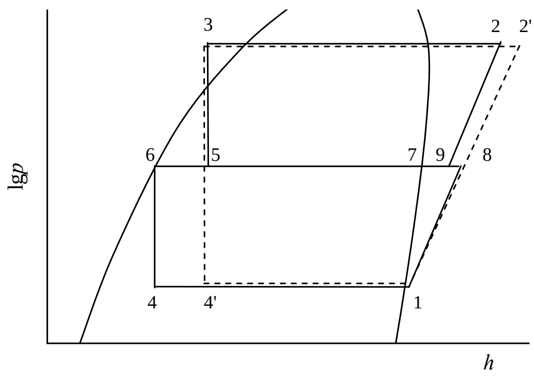


Figure 3: p-h diagram for two-stage vapor injection cycle with flash tank

2.2 Thermodynamic Model for Two-stage Vapor Injection Cycle with Flash Tank

Mass conservation in the flash tank can be expressed as

$$M_c = M_e + M_i \tag{1}$$

The refrigerant quality in flash tank is

$$x_{FT} = \frac{h_{c,out} - h_{FT,l}}{h_{FT,g} - h_{FT,l}} \tag{2}$$

Assuming the saturated vapor in flash tank is fully separated from the saturated liquid and is injected, then we can have

$$M_i = M_c x_{FT} \tag{3}$$

$$M_e = M_c\left(1 - x_{FT}\right) \tag{4}$$

Heating capacity is

$$Q_c = M_c\,(h_{c,in} - h_{c,out}) \tag{5}$$

The enthalpy of discharge refrigerant from low-stage compressor is

$$h_{dis,LS} = h_{suc,LS} + \frac{h_{dis,LS}^{is} - h_{suc,LS}}{\eta_{is,LS}} \tag{6}$$

The compression work of low-stage compressor is

$$W_{LS} = M_e(h_{dis,LS} - h_{suc,LS}) \tag{7}$$

Energy conservation in the mixing process can be expressed as

$$M_c h_{suc,HS} = M_e h_{dis,LS} + M_i h_{FT,g} \tag{8}$$

Combined equation (3) and equation (4) into equation (8),

$$h_{suc,HS} = \left(1 - x_{FT}\right)h_{dis,LS} + x_{FT}h_{FT,g} \tag{9}$$

The enthalpy of discharge refrigerant from high-stage compressor is

$$h_{dis,HS} = h_{suc,HS} + \frac{h_{dis,HS}^{is} - h_{suc,HS}}{\eta_{is,HS}} \tag{10}$$

The compression work of high-stage compressor is

$$W_{HS} = M_c\,(h_{dis,HS} - h_{suc,HS}) \tag{11}$$

The total compression work is

$$W = W_{HS} + W_{LS} \tag{12}$$

The mass flow rate through evaporator and condenser can be expressed as

$$M_e = \eta_{v,LS}\frac{V_{rev,LS}f_{LS}}{v_{suc,LS}} \tag{13}$$

$$M_c = \eta_{v,HS}\frac{V_{rev,HS}f_{HS}}{v_{suc,HS}} \tag{14}$$

The displacement ratio of high-stage to low-stage is reduced from equation (4), equation (13) and equation (14)

$$R_v = \frac{f_{HS}}{f_{LS}}\frac{V_{rev,HS}}{V_{rev,LS}} = \frac{\eta_{v,LS}}{\eta_{v,HS}}\frac{v_{suc,HS}}{v_{suc,LS}}\frac{1}{\left(1 - x_{FT}\right)} \tag{15}$$

COP and volumetric heating capacity of the two-stage vapor injection cycle are

$$COP = \frac{Q_c}{W} \tag{16}$$

$$q_{v,c} = \frac{Q_c}{\eta_{v,LS}V_{rev,LS}f_{LS}} \tag{17}$$

The thermodynamic model can be solved through EES (Engineering Equation Solver, Klein S. A. & Alvarado F. L., 2002) when the refrigerant states of low-stage suction vapor, high-stage discharge vapor, outlet liquid of condenser are specified as well as isentropic efficiency and volumetric efficiency of each stage compressor.

2.3 Theoretical Optimum Displacement Ratio Analysis

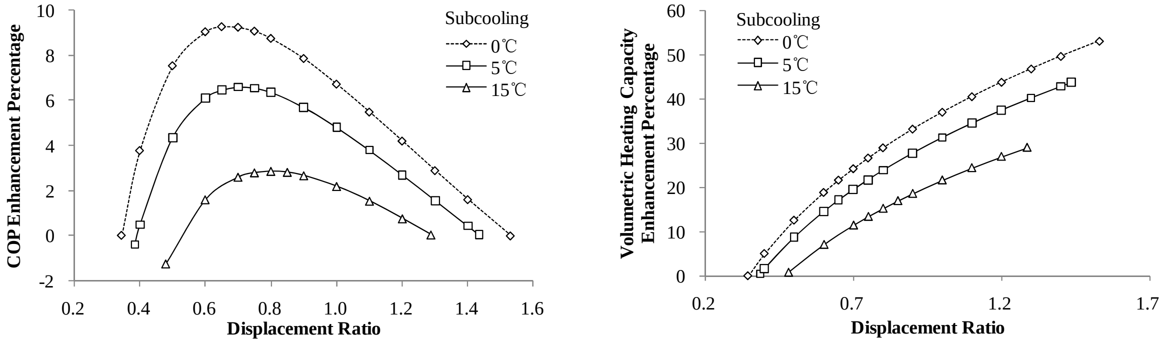
The calculated thermodynamic conditions are as following:

(1) Isentropic efficiency of each stage is 0.7
(2) Outlet liquid subcooling of condenser is 0℃~15℃
(3) Superheat of suction vapor is 0℃
(4) Condensation temperature is 25℃~65℃
(5) Evaporation temperature is -40℃~20℃

The volumetric efficiency of each stage is assumed the same, 0.85 if needed.

As is shown in figure 4 (a), COP of R410A two-stage vapor injection cycle is improved up to 9.3% as compared to one stage cycle when evaporation temperature, condensation temperature and subcooling are 0℃, 45 ℃ and 0℃, respectively. There exists an optimum displacement ratio corresponding to the maximum COP. The maximum COP decreases and the optimum displacement ratio increases with subcooling increasing.

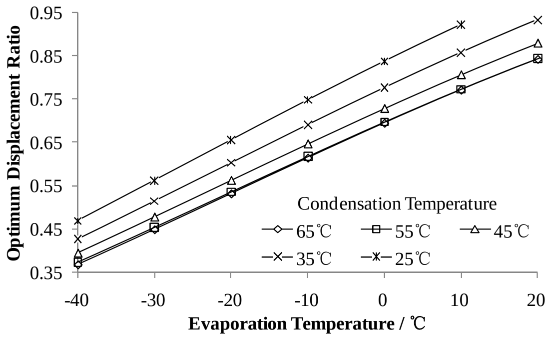
As is shown in figure 4 (b), the volumetric heating capacity of R410A two-stage vapor injection cycle is improved up to 53% as compared to one stage cycle when evaporation temperature, condensation temperature and subcooling are 0℃, 45 ℃ and 0℃, respectively. The volumetric heating capacity enhancement percentage increases with displacement ratio increasing and decreases with subcooling increasing. As can be seen in figure 3, the refrigerant quality in flash tank will decrease with subcooling increasing and will increase with medium pressure in flash tank decreasing. The volumetric heating capacity enhancement percentage will increase when the quality in flash tank increases. When displacement ratio increases, the medium pressure will decrease to balance mass conservation. Consequently, quality will increase and thus volumetric heating capacity enhancement percentage increases.



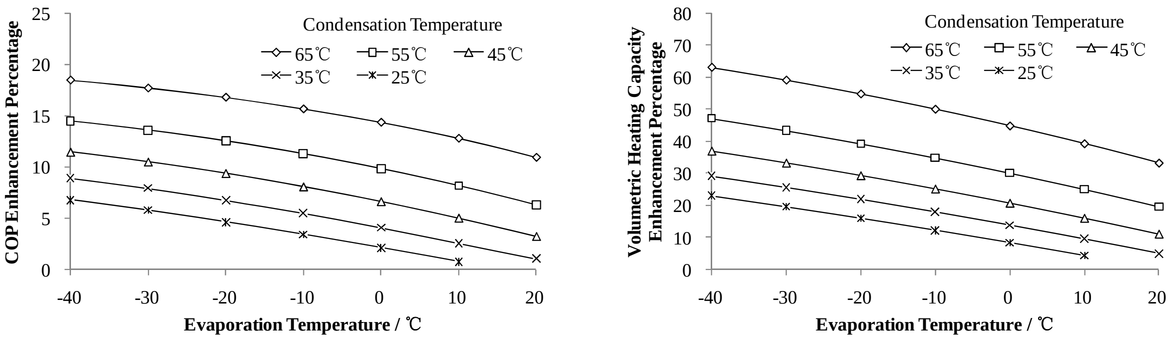
(a) COP enhancement (b) Volumetric heating capacity enhancement

(R410A, Evaporation temperature = 0℃, Condensation temperature = 45℃)

Figure 4: COP and volumetric heating capacity enhancement percentage relative to single stage



(a) Optimum displacement ratio



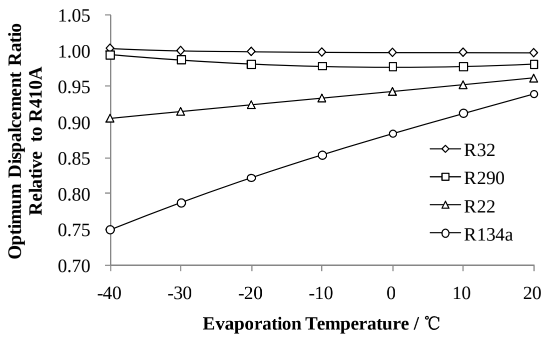
(b) COP enhancement (c) Volumetric heating capacity enhancement

(R410A, Subcooling=5℃)

Figure 5: Optimum displacement ratio, corresponding COP and volumetric heating capacity enhancement percentage relative to single stage

As is shown in figure 5(a), the R410A theoretical optimum displacement ratio decreases almost linearly with evaporation temperature decreasing, and decreases with condensation temperature increasing, when the subcooling is 5℃. As is shown in figure 5(b) and figure 5(c), the maximum COP improvement and volumetric heating capacity improvement both increase with evaporation temperature decreasing and condensation temperature increasing, and can be up to 18.5% and 63%, respectively, under the calculated thermodynamic conditions for R410A.

The theoretical optimum displacement ratio of R32, R290, R22 and R134a relative to that of R410A is shown in Figure 6 when condensation temperature and subcooling are 45℃ and 5℃, respectively. The optimum displacement ratio of R410A, R32 and R290 is very close to each other under the calculated evaporation temperature range from -40℃ to 20℃. The relative displacement ratio of R22 and R134a is 0.905~0.962 and 0.749~0.939, respectively.



(Condensation temperature = 45℃, Subcooling=5℃)

Figure 6: Relative Optimum displacement ratio of R32, R290, R22 and R134a to that of R410A

The optimum displacement ratio correlation for R290, R32 and R410A under the calculated conditions is suggested as

$$R_{v,opt} = \left(c_0 + c_1 t_e + c_2 t_c + c_3 t_e t_c + c_4 t_e^2 + c_5 t_c^2\right)\left(c_6 + c_7\, \Delta t_{sc}/t_c\right) \tag{18}$$

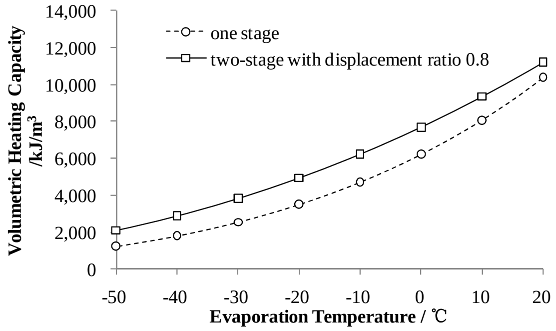
Coefficients of the optimum displacement ratio correlation are listed in table 1. The mean and maximum relative deviation between the predicted optimum displacement ratio and the theoretical optimum displacement ratio is within 0.8% and 4% for R290, respectively, under the calculated conditions.

Table 1: Coefficients of equation (18)

c_0	c_1	c_2	c_3	c_4	c_5	c_6	c_7
9.8596E-01	9.3684E-03	-9.1383E-03	-3.9061E-05	-6.0952E-06	5.6071E-05	9.9995E-01	2.8089E-01

2.4 Variable Displacement Ratio for Heating Capacity Enhancement Analysis

As is shown in figure 7, the calculated R410A volumetric heating capacity for one stage cycle and two-stage vapor injection cycle (displacement ratio=0.8) both decreases with evaporation decreasing. The heating capacity of two-stage vapor injection cycle (displacement ratio=0.8) will be improved 11%~40% (or higher if take into account the volumetric efficiency improvement) as compared to one stage cycle under the same calculated conditions, when the compressor displacement of the two cycles is the same. The heating capacity enhancement ratio (1.11~1.4) is not enough for low temperature ASHP in cold climate region in practice even if the inverter-driven compressor used.

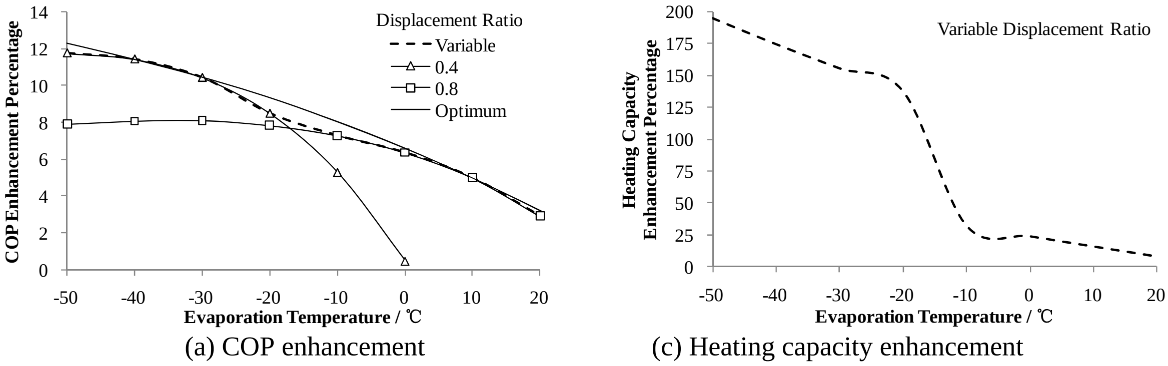


(Condensation temperature = 45℃, Subcooling=5℃)

Figure 7: R410A volumetric heating capacity of singe stage and two stage cycle

The heating capacity will be improved if the compressor displacement (including inverter-driven compressor's frequency) increases. The displacement of low-stage compressor should be enlarged in lower evaporation temperature to enhance heating capacity as well as COP in two-stage vapor injection cycle, which is in harmony with what we need in practice and in theory. To avoid the oil return risk and reduce the compressor unit cost, Liang (2014) designed a new type two-stage vapor injection compressor with variable displacement ratio for heating capacity enhancement as well as COP enhancement used in cold climate region. The two-stage vapor injection triple-cylinder rotary compressor has two cylinders in low stage and one cylinder in high stage, and one of the low-stage cylinder's operating state can be switched on or off to enlarge or shrink the low-stage operating displacement corresponding to low or medium ambient temperature.

As is shown in figure 8, the theoretical COP improvement (dashed line in the figure) of the new type two-stage vapor injection cycle with variable displacement ratio (0.8 &0.4) is very close to the theoretical maximum COP improvement (real line without data mark in the figure) in the evaporation temperature range -50℃~20℃. The COP and heating capacity can be improved up to 10.5% and 155.6%, respectively, as compared to one-stage cycle with the same cylinder displacement as one of the low-stage cylinder displacement, when evaporation temperature, condensation temperature and subcooling is -30℃, 45℃ and 5℃, respectively.



(a) COP enhancement (c) Heating capacity enhancement

(Condensation temperature = 45℃, Subcooling=5℃)

Figure 8: COP and heating capacity enhancement percentage for R410A two-stage vapor injection cycle with variable displacement ratio

The two displacement ratios of the new type two-stage vapor injection compressor with variable displacement ratio should be optimized based on the operating conditions in practice, compressor's volumetric efficiency, seasonal energy efficiency ratio (SEER) or annual performance factor (APF) and so on.

3. EXPERIMENT RESULTS

As is shown in figure 9, a new prototype of two-stage vapor injection air source heat pump (ASHP) with a variable displacement ratio rotary compressor was designed and manufactured for experiment. The compressor of the new type ASHP is an inverter-driven two-stage triple-cylinder rotary compressor with two cylinders in low stage and one cylinder in high stage, total displacement of low stage is 51.8 cm^3/rev. The condenser and evaporator of the tested ASHP were fin-and-tube heat exchangers, which were used for 13.5kW (rated heating capacity under inside room 20℃/15℃ and outside room 7℃/6℃) one stage ASHP. An inverter-driven one-stage rotary compressor with displacement 30.6 cm^3/rev was used for comparison. The refrigerant is R410A.

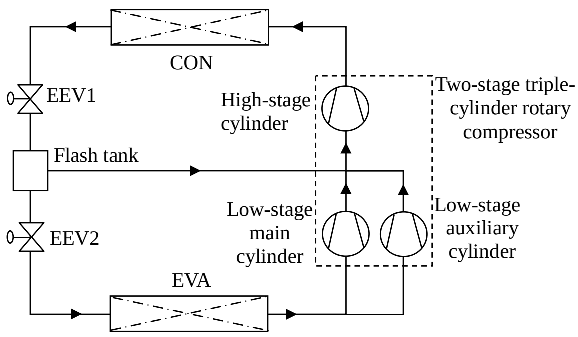


Figure 9: R410A two-stage vapor injection heat pump with a variable displacement ratio rotary compressor

Heating capacity, power consuming, suction / discharge pressure, etc. were tested in the conventional Air Enthalpy Method Calorimeter lab. The operating conditions for heating listed in table 2.

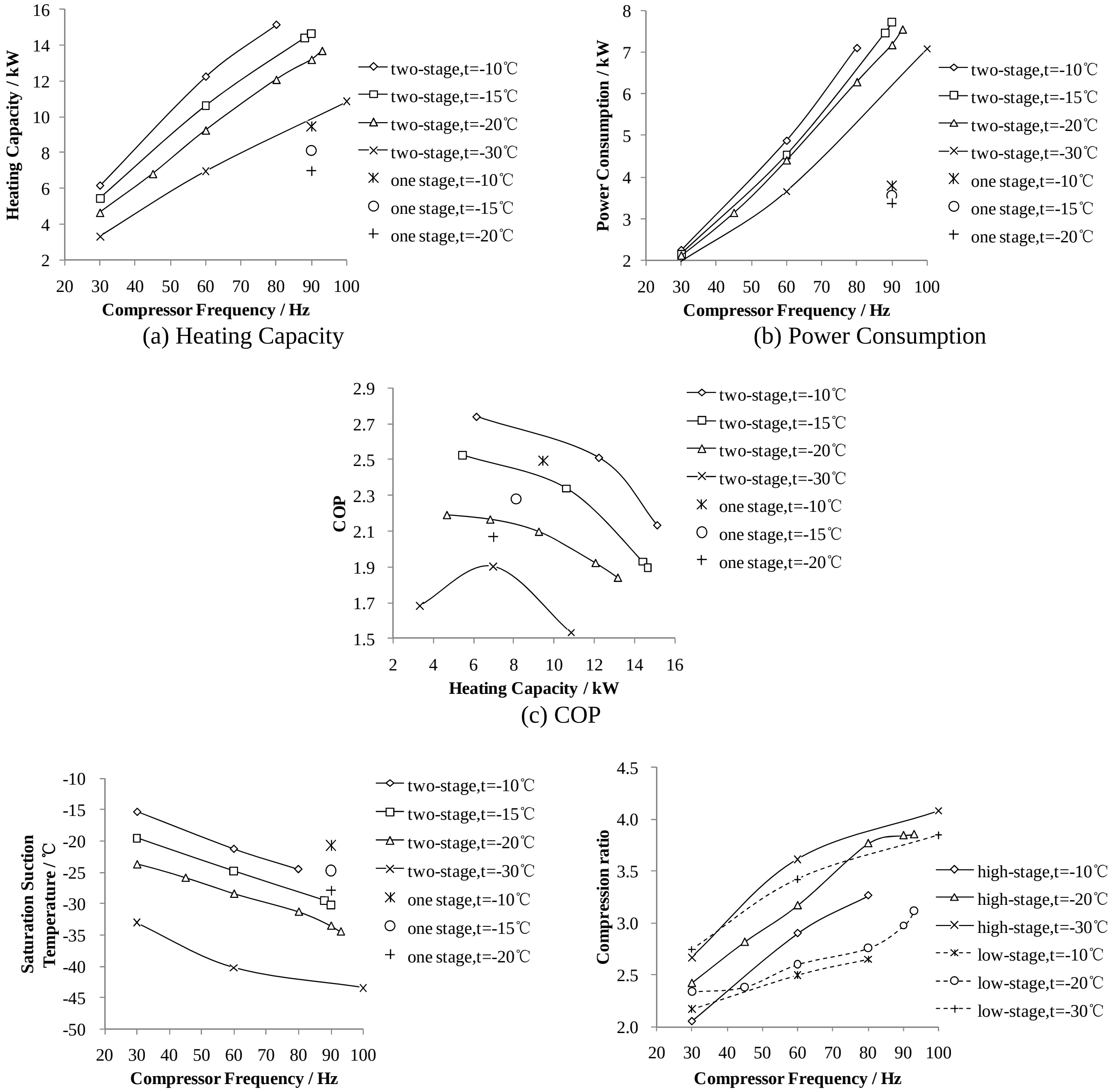
Table 2: Operating conditions for heating (unit: ℃)

	Inside room	Outside room
Dry bulb temperature	20	-10、-15、-20、-30
Wet bulb temperature	15	-

The experiment results are shown in figure 10. As is shown in figure 10 (a) and figure 10(b), heating capacity and power consumption of the new type ASHP both increase with compressor frequency increasing and decrease with outside room temperature decreasing, and are remarkably higher than those of conventional ASHP under same operation conditions. Heating capacity of the new type ASHP can be over 13.7kW (higher than rated heating capacity 13.5kW) when the outside room temperature is over -20℃, and can be over 10.8kW (80% higher than rated heating capacity 13.5kW) when the outside room temperature is -30℃. Heating capacity of the new type ASHP is improved 96% as compared to the conventional ASHP when the outside room temperature is -20℃.

As is shown in figure 10 (c), COP of the new type ASHP is improved 5%~10% when heating capacity is comparable to the conventional ASHP. Heating capacity of the new type ASHP could improve 30%~50% when COP was comparable to the conventional ASHP.

As is shown in figure 10 (d), the saturation suction temperature corresponding to the suction pressure of the new type ASHP decreases with compressor frequency increasing, and is evidently lower than that of conventional ASHP under the same operation conditions.



(a) Heating Capacity (b) Power Consumption

(c) COP

(d) Saturation suction temperature (e) Compressor ratio of high stage and low stage

Figure 10: Experiment results

As is shown in figure 10 (e), the compression ratio of high stage and low stage both increase with compressor frequency increasing, and is in the range of 2.0~4.1 and 2.2~3.8, respectively, thus the total compression ratio is in the range of 4.4~15.6.

4. CONCLUSIONS

In this paper, two-stage vapor injection compression cycle with flash tank was thermodynamically analyzed, and the optimum theoretical displacement ratio correlation for R290, R32 and R410A was given. A new type two-stage vapor injection low temperature ASHP with a variable displacement ratio rotary compressor was designed and tested.

Thermodynamic analysis results show that the theoretical maximum COP improvement and volumetric heating capacity improvement of two-stage vapor injection cycle to one stage cycle both increase with evaporation temperature decreasing and condensation temperature increasing, and can be up to 18.5% and 63%, respectively, when evaporation temperature, condensation temperature and subcooling is -40℃, 65℃ and 5℃, respectively. The theoretical COP and heating capacity of the new type two-stage vapor injection compressor with variable displacement ratio can be improved up to 10.5% and 155.6%, respectively, when evaporation temperature, condensation temperature and subcooling is -30℃, 45℃ and 5℃, respectively.

The experimental results of the new type ASHP showed that the heating capacity under 20℃/-20℃(inside room /outside room) could reach the rated heating capacity under 20℃/7℃, improving 96% compared to the conventional one-stage ASHP, the heating capacity under 20℃/-30℃ could reach 80% of the rated one. COP of the new type ASHP could improve 5%~10% when the heating capacity was comparable to the conventional ASHP, and the heating capacity of the new type ASHP could improve 30%~50% when COP was comparable to the conventional ASHP.

NOMENCLATURE

COP	coefficient of performance	dis	discharge
f	frequency	e	evaporator or evaporation
M	mass flow rate	FT	flash tank
h	enthalpy	g	gas
Q	heating capacity	HS	high stage
q	volumetric heating capacity	i	injection
R	displacement ratio	in	inlet
t	temperature	is	isentropic
V	displacement	l	liquid
v	specific volume	LS	low stage
W	compression work	out	outlet
x	quality	sc	subcooling
η	efficiency	suc	suction
		sat	saturation
Subscript		v	volume
c	condenser or condensation		

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